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NUMERICAL INVESTIGATION OF THE TRANSIENT CONDUCTION IN A ROTATING CYLINDRICAL SHELL EXPOSED TO AN INCIDENT TIME VARYING HEAT FLUX

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ABSTRACT

The transient temperature distribution in a rotating cylindrical shell which is heated by an incident time varying heat flux (nuclear pulse) as well as a constant heat flux, is determined numerically by a finite difference method. The shell is cooled by combined convection and thermal radiation. The effects of cooling and rotation on the temperature distribution as well as the time- and space-dependence are shown. Rotation provides a sinusoidal temperature variation in time for a fixed surface and circumferential position. Increased rotation reduces the maximum temperature in the shell and also provides a more uniform temperature distribution in the circumferential direction.

KEY WORDS Transient temperature distribution Rotating cylindrical shell Incident time ranging heat flux

NOMENCLATURE

$AN1,j$ = part of coefficient
A,B,C = constants
a_1 = thermal diffusivity, $k/\rho c_p$
$a_{i,j}, b_{i,j}, c_{i,j}, d_{i,j}$ = coefficients in the numerical solution
Bi = Biot number, $h(r_2 - r_1)/k$
c_p = specific heat
D = duration of the incident heat pulse
D = outer diameter
dt' = dimensionless time step
Fo = Fourier number
Gr = Grashof number
g = gravitational acceleration
h = convective heat transfer coefficient
i,j = indices for grids
k = thermal conductivity in the shell
k_f = thermal conductivity of air
NHF = heat flux number, $\alpha \dot{Q}_{max}(r_2 - r_1)/kT_{init}$
N,M = number of grid points
NR = radiation number, $\sigma\varepsilon T^3_{init}(r_2 - r_1)/k$
Nu = Nusselt number
n = time level
Pr = Prandtl number
$\dot{Q}$ = incident heat flux
$\dot{Q}_{in}$ = dimensionless heat flux, $\dot{Q}/\dot{Q}_{max}$
$\dot{Q}_{max}$ = maximum value of $\dot{Q}$
R' = $r' + r_1/(r_2 - r_1)$
Re, Re_{ω} = Reynolds number
r = radial co-ordinate
r_1 = inner radius of the solid shell
r_2 = outer radius of the solid shell
r' = dimensionless radial co-ordinate, $(r - r_1)/(r_2 - r_1)$
s = subscript denotes outer surface
T = temperature
T' = dimensionless temperature, $(T\text{-}T_{init}/T_{init})$
T_{init} = initial temperature
T_{01} = radiation background temperature
T_{02} = ambient air temperature
T_s = surface temperature
t = time
t' = dimensionless time, $a_1 t/(r_2 - r_1)^2$, Fourier number
u_x = free stream velocity

Greek symbols
α = absorptivity for the incident radiative heat flux
β = volumetric thermal expansion coefficient
γ = angle
Δ = step size
ε = emissivity for thermal radiation
θ = angle co-ordinate
ν = kinematic viscosity
ρ = density
σ = Stefan-Boltzmann constant
φ = dimensionless angle, $\theta/2\pi$
φ_0 = original position
ω = angular velocity rad/s
Ω = dimensionless angular velocity $(\Omega/(2\pi))(r_2 - r_1)^2/a_1$

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INTRODUCTION

Transient heat conduction in a solid with convective or radiative cooling at the surface occurs frequently in real world applications. This paper is concerned with transient heat conduction in a rotating cylindrical shell exposed to a time varying incident heat flux as well as a constant heat flux and the outer surface is cooled by thermal radiation and convection.

With the space adventure in the 1960s a number of papers were published concerning interplanetary conditions. One of those more related to this paper is that of Charnes and Raynor[1] where analytical solutions of the temperature distribution in a thin walled rotating cylindrical body exposed to solar heating were presented. The study was continued a few years later by Olmstead *et al.*[2] by adding a time varying incident heat pulse. However, only conduction in the circumferential direction was considered and no convective loss was considered. More recently Sundén[3] analysed a problem similar to the one in this paper. The most important difference was that in that paper the cylindrical shell did not rotate. A brief literature survey was also presented there. The present work may be regarded as an extension and continuation of the work by Sundén[3].

FORMULATION OF THE PROBLEM

A homogeneous circular cylindrical shell, shown in *Figure 1*, with the inner surface thermally insulated, is rotating uniformly around its axis with the angular velocity ω. Initially the shell has a uniform temperature T_{init}. Suddenly the outer surface is exposed to a time dependent incident radiative heat flux as depicted in *Figure 2*. The incident heat flux is assumed to be an optically parallel beam, which means that only the surface facing the incident flux is exposed, and that at an arbitrary position on the surface the heat flux should be multiplied by the cosine of the angle between the surface normal and direction of the heat flux. Cooling takes place at the outer surface by convection with the ambient air at temperature T_{02}, and by thermal radiation exchange with the background at a temperature T_{01}.

The following assumptions are applied in the analysis: two dimensional heat conduction in space, the cylinder material is opaque to thermal radiation, no heat sinks or sources are present, the physical properties and the convective heat transfer coefficient are uniform and independent of time and temperature, the ambient air is transparent to thermal radiation, the ambient and background temperatures are independent of time.

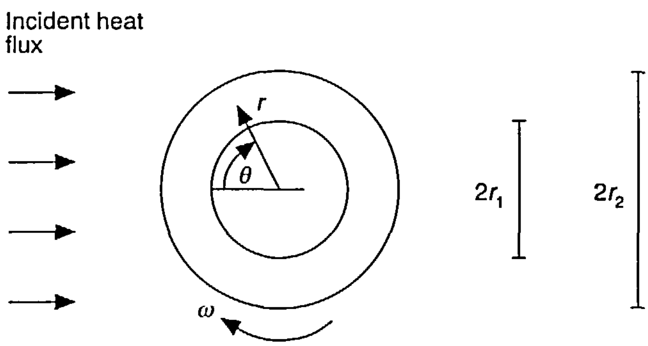

Figure 1 Problem under consideration

THE INCIDENT HEAT FLUX

In engineering applications, the incident heat flux can be of various types and in Schneider[4] some examples are provided. Most of these are for aerospace applications occurring when ascending or entering the planetary atmosphere. Other examples of transient heat fluxes are radiative heat from a nuclear burst, solar flux and radiant heating from a laser beam. The incident heat fluxes considered in this paper are a heat pulse like that of a nuclear bomb and a constant heat flux like solar flux. Data of the transient heat flux released from a nuclear burst have been curve-fitted by Olmstead *et al.*[2] but the exponential expression for that curve has some shortcomings for small values of t (time) and is not suitable for this work. Instead a heat pulse which is assumed piecewise linear with time is adapted from Sundén[3], see *Figure 2*. The total heat energy from the pulse is: $Q_{tot} = 0.1262 Q_{max} D$ where D is the duration time of the pulse.

GOVERNING EQUATION AND BOUNDARY CONDITIONS

The heat conduction equation in two dimensional cylindrical co-ordinates with a constant thermal conductivity is transformed into a non-dimensional form by introducing the following dimensionless variables,

$$T' = (T - T_{init})/T_{init},\ r' = (r - r_1)/(r_2 - r_1),\ \varphi = \theta/2\pi,\ t = a_1 t/(r_2 - r_1)^2$$

One then obtains,

$$\frac{\partial T'}{\partial t'} = \frac{\partial^2 T'}{\partial r'^2} + \frac{1}{R'}\frac{\partial T'}{\partial r'} + \frac{1}{(2\pi R')^2}\frac{\partial^2 T'}{\partial \varphi^2} \qquad (1)$$

where $R' = r' + r_1/(r_2 - r_1)$.

The shell is at a uniform temperature initially. The initial condition of the problem is then,

$$T'(r', \varphi, 0) = 0. \qquad (2)$$

The inner surface of the shell is thermally insulated and the boundary condition is thus,

$$\partial T'/\partial r'(0, \varphi, t') = 0. \qquad (3)$$

For the outer surface, where the front surface is exposed to the incident radiative heat flux, and heat exchange by thermal radiation and convection also occurs, one obtains in non-dimensional form,

$$\frac{\partial T'}{\partial r'_s} = \frac{(r_2 - r_1)}{k}\left[\frac{\alpha \dot{Q}\cos(\gamma)}{T_{init}} - h(T'_s - T'_{02}) - \sigma\varepsilon T^3_{init}\left((T'_s + 1)^4 - (T'_{01} + 1)^4\right)\right]. \qquad (4)$$

In (4) subscript s denotes surface and γ is the angle between the surface normal and the incident heat flux. The angle θ is measured from a fixed point. With θ measured as in *Figure 1* one has $\gamma = \theta + \omega t$. Only positions with $\cos(\gamma) \geq 0$ will receive the incident heat flux, i.e. on the side exposed to the incident flux. Since the shell is rotating with a constant speed, the surface elements will be exposed in a periodic manner. It should be noted that if no cooling is present, the incident heat flux is totally absorbed.

Two well-known non-dimensional numbers are recognized in (4), the Biot number $Bi = h(r_2 - r_1)/k$ and the radiation number $NR = \sigma\varepsilon T^3_{init}(r_2 - r_1)/k$. By defining a heat flux number as $NHF = \alpha\dot{Q}_{max}(r_2 - r_1)/kT_{init}$, (4) can be written as,

$$\frac{\partial T'}{\partial r'_s} = Q_{in} NHF\cos(\gamma) - Bi(T'_s - T'_{02}) - NR((T'_s + 1)^4 - (T'_{01} + 1)^4) \qquad (5)$$

where $\dot{Q}_{in} = \dot{Q}(t)/\dot{Q}_{max}$.

By using various values of *NR* and *Bi*, the importance of the radiative and convective cooling can be investigated. The strength of the incident heat flux is controlled by the heat flux number *NHF*. The rotation affects the heat transfer process primarily by controlling the exposure time periods of the surface. Its influence on the convective cooling rate is discussed in another section.

NUMERICAL APPROXIMATION

By using Taylor series expansions and replacing space derivatives by central finite difference approximations, and the time derivative by a first order forward approximation and using constant grid spacing in each direction, (1) becomes,

$$\frac{T_{i,j}'^{n+1} - T_{i,j}'^{n}}{\Delta t'} = \frac{T_{i+1,j}' + T_{i+1,j}' - 2T_{i,j}'}{\Delta r'^2} + \frac{T_{i+1,j}' + T_{i-1,j}'}{2\Delta r' R'} + \frac{T_{i,j+1}' + T_{i,j-1}' - 2T_{i,j}'}{(2\pi R' \Delta\varphi)^2} \tag{6}$$

where i represents the r-direction, j the φ-direction and n denotes the time level.
By taking an energy balance for a control volume situated at the outer surface[5], one obtains in non-dimensional form,

$$\Delta r' R_{N-1/4}' (T_{N,j}'^{n+1} - T_{N,j}'^{n}) / 2\Delta t' = R_{N-1/2}' (T_{N-1,j}' - T_{N,j}') / \Delta r'$$

$$+ \Delta r' (T_{N,j-1}' + T_{N,j+1}' - 2T_{N,j}') / 2R_N' (\Delta\varphi 2\pi)^2 + R_N' \frac{\partial T'}{\partial r_s'} \tag{7}$$

where $\partial T'/\partial r'_s$ is the boundary condition for the outer surface according to (5). From (7) the required expression for the temperature at the surface grid point ($T_{N,j}$) can be obtained. Index N denotes the outer surface, so that $R'_{N-1/4}$ means $R'_N - \Delta r'/4$.

If the temperatures on the right hand side of (6) and (7) are taken at time level n, i.e. the temperatures are known, one obtains an explicit form and the unknown temperatures at time level $n + 1$ can be solved directly. If the temperatures on the right hand side are taken at time level n + 1 (unknown), one obtains an implicit form and the equations must be solved iteratively or simultaneously at each time level. The explicit form is however sometimes unstable and hence the implicit form is used.

In order to reduce the necessary number of nodes in the r-direction and enhance the numerical procedure, the temperature in the energy storage term (the left-hand side of (7)) is set at an additional nodal point N-1/4, i.e. in the centre of the control volume. By using the one dimensional heat conduction equation the temperature in the control volume can be expressed in terms of the neighbouring values as,

$$T_{N-1/4,j}' = \frac{\ln(A)}{\ln(B)} T_{N,j}' + \frac{\ln(C)}{\ln(B)} T_{N-1,j}' \tag{8}$$

where $A = R'_{N-1/4,j}/R'_{N-1,j}$, $B = R'_{N,j}/R'_{N-1,j}$, $C = R'_{N,j}/R'_{N-1/4,j}$.

METHOD OF NUMERICAL SOLUTION

Equations (6) and (7) are set in implicit forms. The properties on the right hand side of the equations are taken at time level n + 1, and due to the non-linear boundary condition the equations are solved iteratively at each time level. For (6) we obtain,

$$a_{i,j} T_{i,j}'^{n+1} = b_{i,j} T_{i-1,j}'^{n+1} + c_{i,j} T_{i-1,j}'^{n+1} + d_{i,j} \tag{9}$$

where

$$a_{i,j} = 1/\Delta t' + 2/\Delta r'^2 + 2/(2\pi R'\Delta\varphi)^2, b_{i,j} = 1/\Delta r'^2 + 1/(2R'\Delta r'),$$

$$c_{i,j} = 1/\Delta r'^2 - 1/(2R'\Delta r'),$$

$$d_{i,j} = \frac{(T'^{n+1}_{i,j-1} + T'^{n+1}_{i,j+1})}{(2\pi R'\Delta\varphi)^2} + \frac{T'^{n}_{i,j}}{\Delta t'}.$$

Equation (7) can be written as,

$$a_{N,j}(T'^{n+1}_{N,j} + 1) = c_{N,j}\, T'^{n+1}_{N-1,j} + d_{N,j} \tag{10}$$

where

$$a_{N,j} = A_{N1,j} + \frac{2R'_N}{\Delta r'} NR(T'^{n+1}_{N,j} + 1)^3$$

$$A_{N1,j} = \frac{R'_{N-1/4}}{\Delta t'}\frac{\ln(A)}{\ln(B)} + \frac{2R'_{N-1/2}}{\Delta r'^2} + \frac{2}{(2\pi\Delta\varphi)^2 R'_N} + \frac{2R'_N}{\Delta r'} Bi$$

$$c_{N,j} = \frac{2R'_{N-1/2}}{\Delta r'^2} - \frac{R'_{N-1/4}}{\Delta t'}\frac{\ln(C)}{\ln(B)}$$

$$d_{N,j} = \frac{R'_{N-1/4}}{\Delta t'}\left(\frac{\ln(A)}{\ln(B)}T^{n}_{N,j} + \frac{\ln(C)}{\ln(B)}T'^{n}_{N-1,j}\right) + \frac{(T'^{n+1}_{N,j+1} + T'^{n+1}_{N,j-1})}{(2\pi\Delta\varphi)^2 R'_N} + A_{N1,j}$$
$$+ \frac{2R'_N}{\Delta r'}\left(\dot{Q}_{in} NHF\cos(\gamma) + NR(T'_{01} + 1)^4 + BiT'_{02}\right).$$

The solution procedure is a line-by-line technique using the TDMA algorithm combined with the Gauss-Seidel method as described by Patankar[6]. For each time-level the procedure starts at the outer surface at $\varphi_0 = \varphi(t = 0) = 0$ solving (10) and then the TDMA method is applied in the radial direction for (9). The procedure is thereafter repeated for the next φ_0 and so on until $\varphi_0 = 1$ is reached and then the procedure restarts at $\varphi_0 = 0$. This iterative process is repeated until a convergence criterion is satisfied which in this case was chosen so that the maximum relative change in all temperatures, between two successive iterations, should be less than 10^{-5}.

All computations were carried out on an IBM compatible PC with a 486 (66 MHz) intel processor.

Sample calculations

The thermal response of the shell depends on the heating and cooling, thermal diffusivity of the shell material, time, size of the shell and the rotation speed. The time and the thermal diffusivity are represented by the Fourier number $Fo = t'$. The cooling is represented by the Biot number Bi and the radiation number NR, while the heating is controlled by the heat flux number NHF.

Calculations have been performed for one shell size (r_1 = 5.9mm, r_2 = 7.3mm), but two different materials were considered, one with a low thermal diffusivity $a_1 = 6.6\cdot10^{-8}$m^2/s (k = 0.15 W/mK) and one with a high thermal diffusivity $a_1 = 6.5\cdot10^{-5}$m^2/s (k = 170 W/mK). The nuclear pulse as shown in *Figure 2* has a duration of 21.4 seconds and reaches its maximum after 1 second. The value of the constant incident flux is chosen to the maximum value of the nuclear pulse. The ambient temperature T_{02}, the background temperature T_{01} and also the initial temperature T_{init} are all set to 288 K. The appropriate number of grid points and the time increment have been determined by comparison with a simple steady state situation and several test calculations. The time increment used for both cases is in real time 10^{-3}s corresponding to a non-dimensional time step $dt' = 3.36\cdot10^{-5}$ and $3.33\cdot10^{-2}$, respectively. For the radial direction N = 16 was appropriate while for the circumferential direction M = 30 was satisfactory.

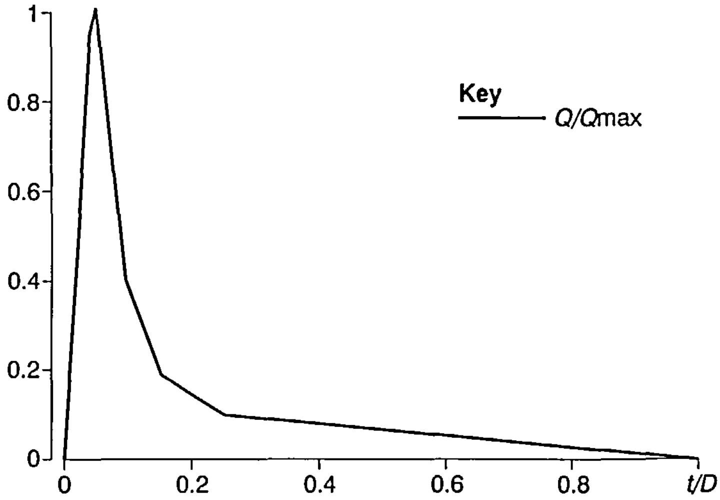

Figure 2 Incident heat pulse vs dimensionless time

The convective heat transfer coefficient is considered constant in time and space and set as an average value. Sundén[3] studied the effects of circumferential variations of the Biot number and found that the effects were very small. To obtain an approximate relevant value of the Biot number a simplified equation for combined effects of rotation, natural convection and forced convection from Wong[7] is applied. The equation reads:

$$\overline{Nu} = 0.135\left[(0.5\, Re_{\omega}^2 + Re^2 + Gr)\, Pr\right]^{0.33} \tag{11}$$

where

$$\overline{Nu} = \frac{hD}{k_f},\ Re_{\omega} = \frac{\omega D^2}{\nu},\ Re = \frac{u_{\infty} D}{\nu},\ Gr = \frac{g\beta D^3 \Delta T}{\nu^2}.$$

The outer shell diameter D is $D = 2r_2$.
Equation (11) is valid if,

$$Re < 1.5\cdot 10^4 \quad 10^3 < Re_{\omega} < 5\cdot 10^4 \quad 0.6 < Pr < 15 \quad \text{value in [ ]} < 10^9.$$

At high free stream velocities the forced convection term dominates. With the fluid properties at an approximate mean value of $T \approx 430$ K and u_{∞}, = 30 m/s, which is considered to be the maximum free stream velocity, equation (11) implies that $Bi_{max} \approx 0.23/k$. Without the forced convection term and with a high rotation speed say $\omega = 100$ rad/s the Biot number will be about one-tenth of Bi_{max}. The maximum radiation number is ($\varepsilon = 1$) $NR_{max} \approx 1.9\cdot 10^{-3}/k$. However, to reveal the effects of the cooling parameters clearly the values of the Biot number and radiation number are extended outside their maximum values in some cases.

RESULTS AND DISCUSSION

Constant incident heat flux

For a constant incident heat flux only the steady conditions are considered. Note that steady conditions mean steady for an outside viewer. However, the temperature for a given point at the shell is changing with the rotation and a given angle φ means different surface elements at different times.

With a thermal diffusivity $a_1 = 6.6{\cdot}10^{-8}\ m^2/s$ the corresponding heat flux number $NHF = 9.1$, and the maximum radiation number $NR_{max} = 0.013$. The circumferential surface temperature distributions for various rotation speeds Ω with NR_{max} and no convective cooling $Bi = 0$, are shown in *Figure 3*. The circumferential variations are greatest for the stationary case and vanish with increasing rotation speed. With higher thermal diffusivity (read conductivity) the variations become smaller and for $k = 170$ W/mK they are less than 5 per cent for the stationary case ($\Omega = 0$). With increased rotation speed the maximum surface temperature moves away from the point facing the incident heat flux ($\varphi = 0$) in the direction of the rotation. At $\Omega \approx 30$ ($\omega = 2\pi$ rad/s) the maximum surface temperature occurs at $\theta \approx 50.4°$ while at infinite speed the maximum will occur at $\theta = 71.46°$[1]. *Figure 4* shows the steady temperature distribution in the shell for $\Omega = 3$. The shell has been scaled-up non-proportionally to improve the interpretation. One can clearly see that the shell is heated when facing the incident heat flux and cooled when it is not. For the stationary case the radial variation is less than 3 per cent. With a high rotation speed, the temperature within the shell becomes homogeneous at a temperature T_s satisfying the heat balance,

$$NHF = \pi\ NR((\bar{T}_s' + 1)^4 - (T_{01}' + 1)^4). \tag{12}$$

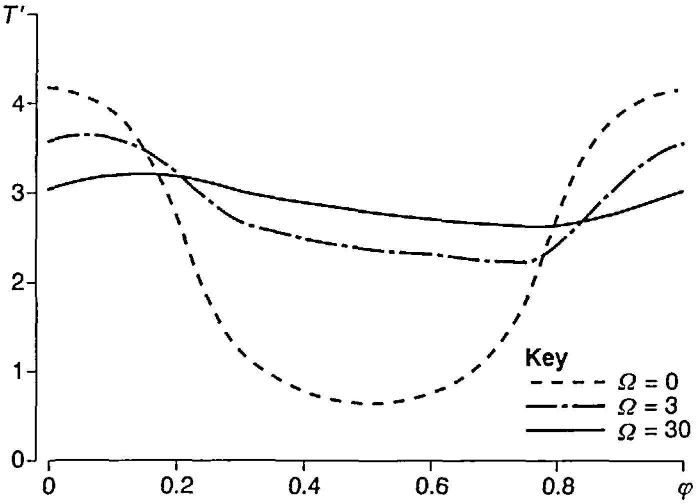

Figure 3 Influence of rotation on the surface temperature with $NR = 0.013$ and $Bi = 0$

The convergence criterion used for the steady condition is that the relative difference for the incoming and reradiated heat is less than 0.5 per cent. Charnes and Raynor[1] present an approximate analytical solution for a thin walled rotating cylinder. However, their solution gives an error which is considerable when the circumferential variations of the temperature are great. For the stationary case presented above the error in the heat balance is more than 50 per cent and thus their result differs considerably from ours. With more moderate circumferential temperature variations the error in their solution becomes much smaller.

Nuclear pulse

Case 1

With a thermal diffusivity $a_1 = 6.6{\cdot}10^{-8}$ m^2/s, the corresponding heat flux number is $NHF = 9.1$. Biot numbers $Bi > 1.5$ and $NR > 0.013$ are of more academic than engineering interest. The dimensionless rotational speed $\Omega = 30$ is equivalent to an angular velocity $\omega = 2\pi$ rad/s.

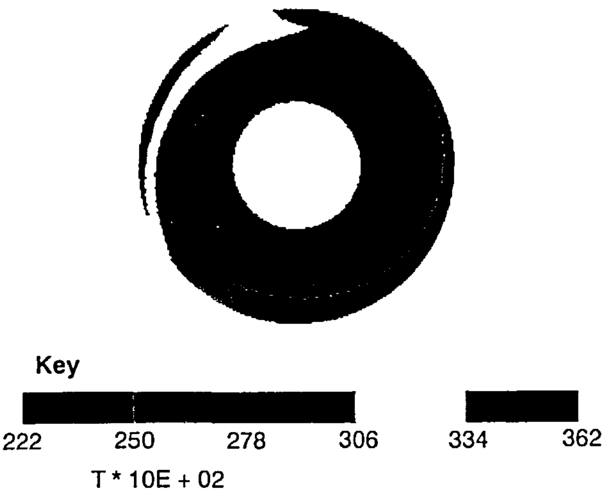

Figure 4 Steady temperature distribution in the shell for $\Omega = 30$ with $NR = 0\,013$ and $Bi = 0$. Please note that the figure is not to scale in the radial direction

Figure 5a shows the surface temperature at $\varphi = 0$ vs dimensionless time, without cooling for various rotation speeds Ω. For small Ω the surface at $\varphi = 0$ is exposed to the incident heat flux for nearly the whole time period of interest. The temperature increases rapidly to its peak value and then decreases as the heat flux decays and the heat is conducted into the body. With higher Ω a surface element is exposed for shorter intervals and the curve becomes more damped. The temperature distributions vs the Fourier number for a given surface point, $\varphi_0 = \varphi(t = 0) = 0$, moving with the rotation speed are shown in *Figure 5b*. Notice the sinusoidal temperature distribution for higher Ω, increasing when exposed to the heat flux and decreasing (the heat is conducted into the body) when not.

Owing to the low thermal diffusivity the heat is slowly conducted into the shell. The radial temperature variation for various times at $\varphi = 0$ and $\Omega = 3$ can be seen in *Figure 6,* and *Figure 7*

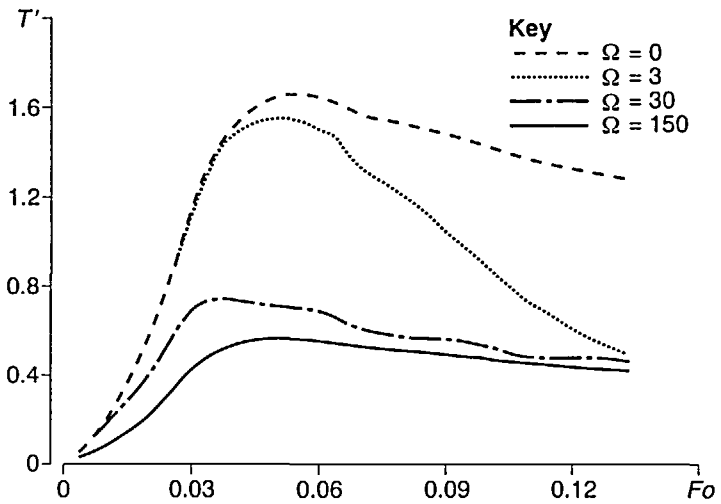

Figure 5a Influence of rotation on the surface temperature at $\varphi = 0$. $Bi = 0$, $NR = 0$

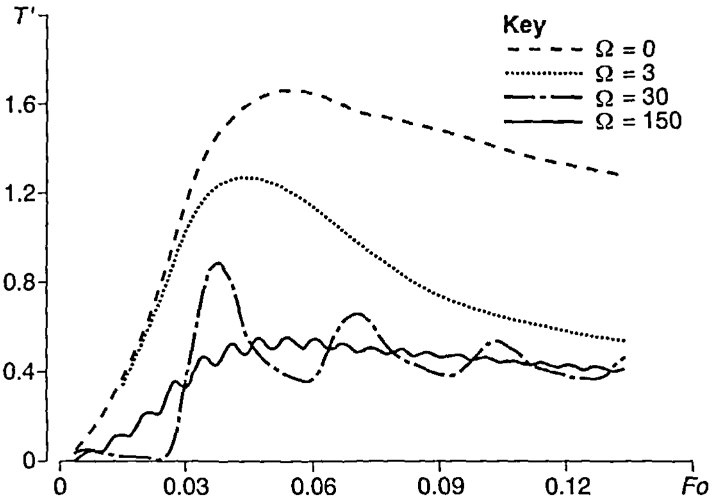

Figure 5b Influence of rotation on the surface temperature at a fixed position moving with the rotation $\varphi(t = 0) = 0. Bi = 0, NR = 0$

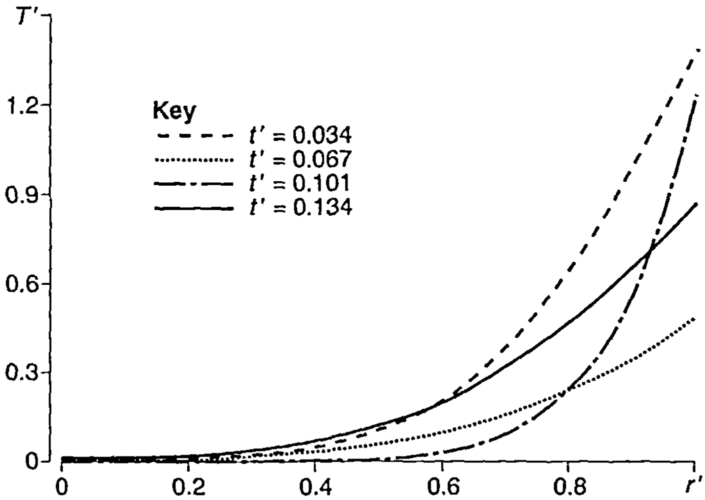

Figure 6 Radial temperature distribution at $\varphi = 0$ for various times with $\Omega = 3$. $Bi = 0, NR = 0$

shows radial distributions for various Ω at $t' = 0.05$ which roughly corresponds to the time when the maximum surface temperature occurs (real time 1.5s). The circumferential variations with the same parameters are shown in *Figures 8* and *9*, respectively. Note the poor heat conduction in the circumferential direction resulting in the fact that the temperature remains at its initial value when not exposed to the incident heat flux. The maximum surface temperature at high Ω is less than half of that at low Ω. As for the case with a constant incident heat flux the maximum temperature moves with the rotation speed and occurs at an angle greater than $\varphi = 0$.

Figure 10 shows the surface temperature response at $\varphi = 0$ for $\Omega = 3$ to the cooling parameters *Bi* and *NR*. When these parameters are increased, the cooling becomes more effective and the temperature level is reduced. The cooling effect of the radiation is at this temperature level about ten times as high as the convective cooling if the Biot number and the radiation number have the

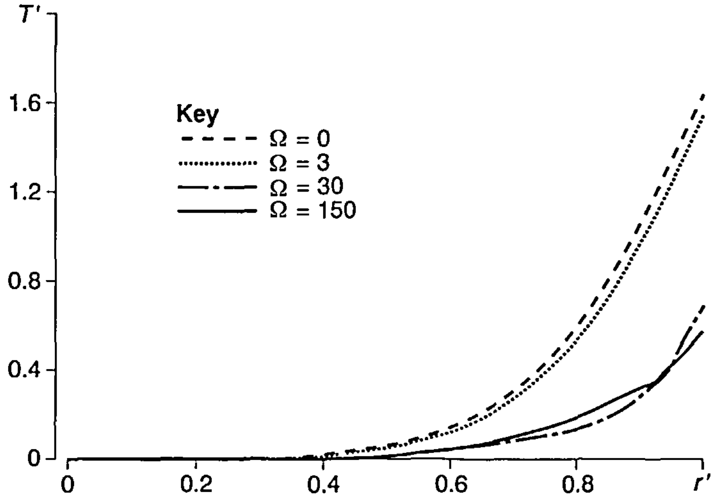

Figure 7 Radial temperature distribution at $\varphi = 0$ for various Θ at $r' = 0.05$. $Bi = 0, NR = 0$

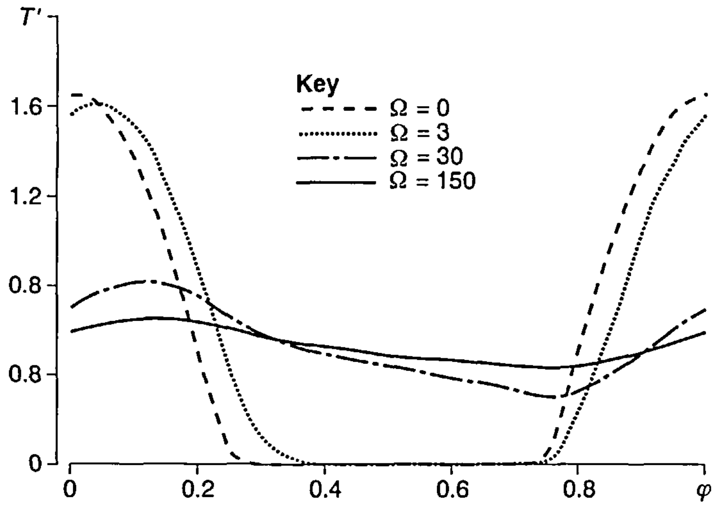

Figure 8 Circumferential surface temperature distribution at various times for $\Omega = 3$. $Bi = 0, NR = 0$

same value. However, as mentioned before, the relevant Biot number is normally much greater than the radiation number, and the curve with $Bi = 1.5$ and $NR = 0$ approximates the real possible maximum cooling effect appropriately. It can also be noticed that the maximum temperature is reached earlier as the values of Bi and NR are increased. Sundén[3] provides a more detailed study of the effects of the cooling parameters.

It seems that rotation is a very important parameter if one wants to reduce the maximum surface temperature. With relevant cooling parameters the maximum surface temperature is reduced by about 25 per cent compared to the non-cooling case, while an increase of the rotation from $\Omega = 3$ to $\Omega = 30$ reduces the maximum temperature by about 50 per cent.

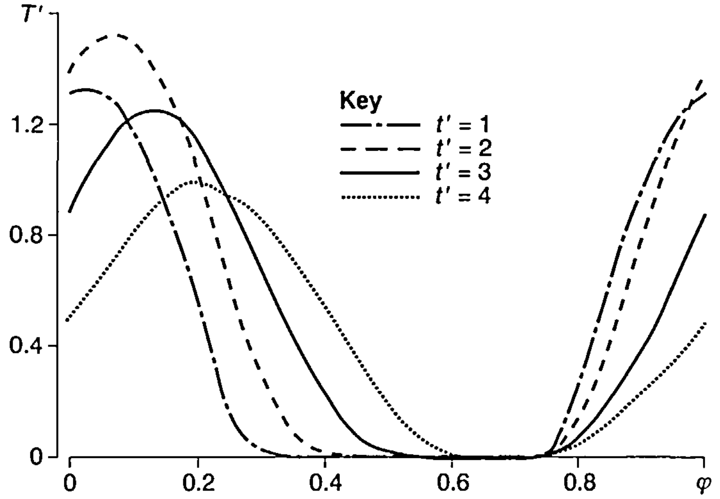

Figure 9 Circumferential surface temperature distribution for various Ω at ' = 0.05. $Bi = 0, NR = 0$

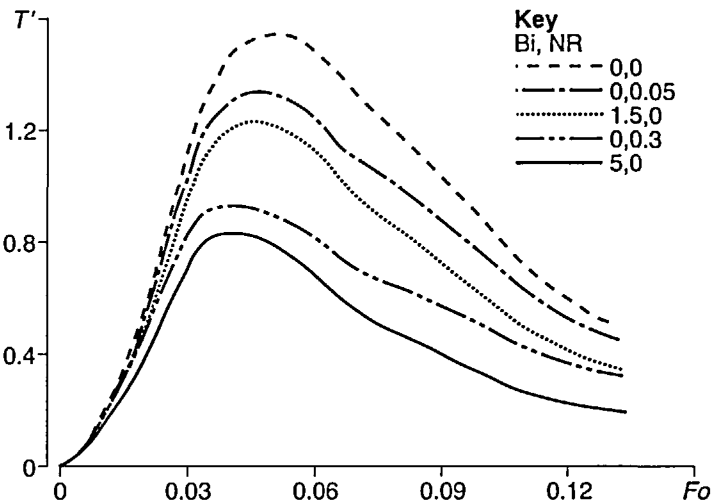

Figure 10 Influence of the convective and radiative cooling parameters for Ω = 3 at $\varphi = 0$

Case 2

A thermal diffusivity $a_1 = 6.5 \cdot 10^{-5}$ m^2/s corresponds to $NHF = 0.0080$ and the relevant Biot and radiation numbers are negligible owing to the high thermal conductivity. Thus only cases without cooling are considered. The rotation speed $\Omega = 0.03$ is equivalent to an angular velocity $\omega = 2\pi$ rad/s. The behaviour of the temperature field due to rotation, cooling and Fourier number (time) is similar to *case 1*, but the temperature level is much lower because of the greater heat conduction within the body. *Figures 11* and *12* provide examples of temperature variations for this case. *Figure 11* shows the temperature distribution in the radial direction at $\varphi = 0$ for $\Omega = 0.003$. A circumferential variation of the surface temperature at $t' = 50$ (real time 1.5 s) is provided in *Figure 12*.

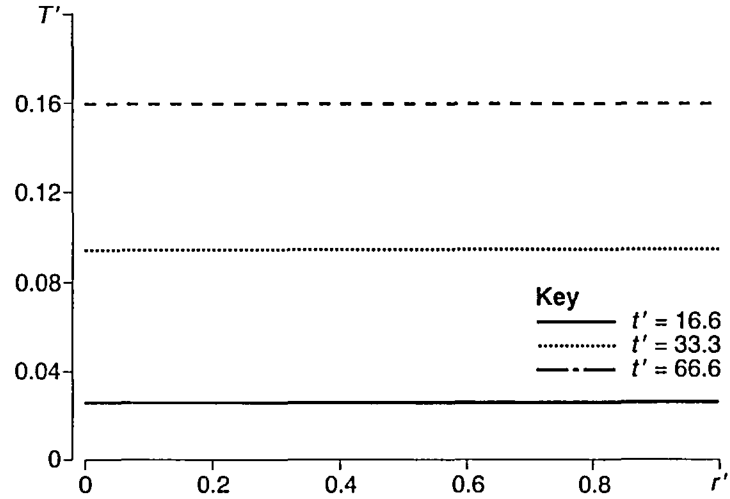

Figure 11 Radial temperature distribution for various times at $\varphi = 0$. $Bi = 0$, $NR = 0$

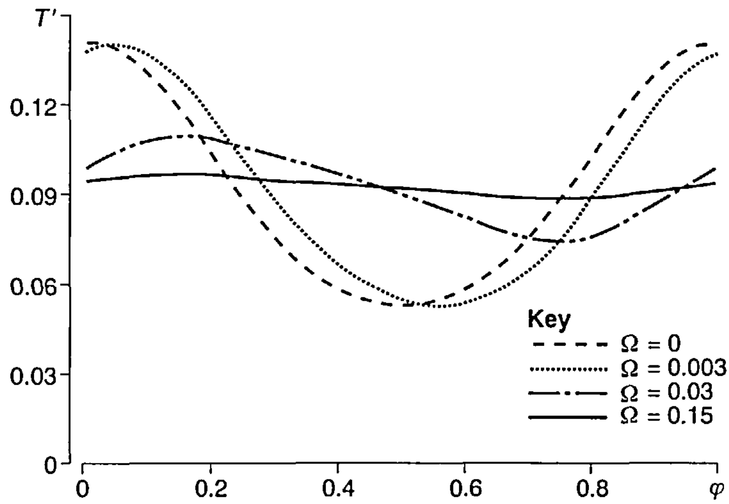

Figure 12 Circumferential surface temperature distribution for various Ω at $t' = 50$. $Bi = 0$, $NR = 0$

CONCLUSIONS

The transient heat conduction in a rotating cylindrical shell exposed to a time varying incident surface heat flux has been numerically investigated. Rotation provides a sinusoidal temperature distribution in time for a given surface and circumferential position. Rotation will also reduce the maximum surface temperature. Increased cooling decreases the temperature level and the maximum temperature is reached earlier. For a shell material with a high thermal diffusivity (conductivity), the temperature levels are much lower and the temperature more evenly distributed within the shell than for a shell material of low thermal diffusivity. For a constant heat flux the steady temperature distribution for the stationary case shows great circumferential variations for a shell material of low thermal conductivity. With higher thermal conductivity or increasing rotation speed the circumferential temperature variations diminish. Rotation will also move the maximum temperature away from the point facing the incident heat flux.

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